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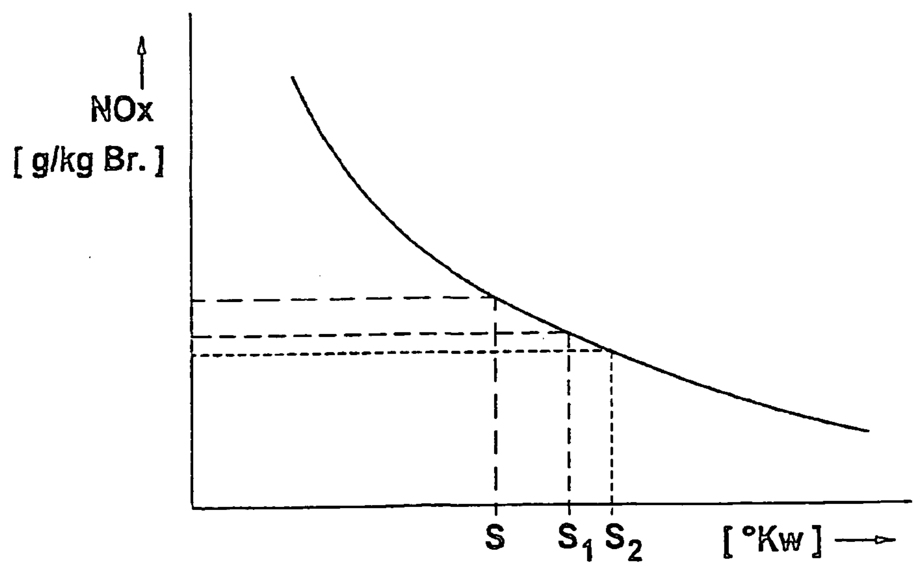
[Fortsetzung auf der nächsten Seite]

(54) Title: METHOD FOR DETERMINING NITROGEN OXIDE CONTENT IN INTERNAL COMBUSTION ENGINE EXHAUST GASES CONTAINING OXYGEN

(54) Bezeichnung: VERFAHREN ZUR BESTIMMUNG DES STICKOXIDGEHALTS IN SAUERSTOFFHALTIGEN ABGASEN VON BRENNKRAFTMASCHINEN



(57) Abstract: During the operation of internal combustion engines, the combustible mixture is compressed in a combustion chamber (11) in at least one cylinder (2) by an alternatingly moveable piston (12). In a method designed to determine the nitrogen oxide content in exhaust gases containing oxygen, the amount of fuel fed to the cylinder (2) and the mass of air flowing in a suction pipe (15) are detected and transmitted to an electric circuit (6). The centre of gravity (S) of combustion is determined from at least one real engine operation measuring value and the amount of nitrogen oxide emission is calculated on the basis of the value for the centre of gravity (S) of combustion, also taking into account the values of the detected amount of fuel and air mass.

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Measurement and Monitoring of Pressure Curves in Diesel Engines

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Abstract

A method for cylinder pressure curve analysis is presented which may be used for real time control and diagnosis of Diesel engines. By subtracting the pressure samples of a firing motor from the non firing case, a difference pressure curve can be obtained which contains hidden information on internal motor conditions. To use this information, a real time data reduction has been implemented which generates symptoms like the centre of gravity or the maximum pressure from the data samples and may be processed by a classification algorithm (i. g. an artificial neural network).

With the presented method it becomes possible to separate heat transfer problems from combustion or injection failures and to adjust motor characteristics properly. The validity of our approach is supported by simulation and experimental data.

1. Introduction

For real time optimization of Diesel engines, for driving at maximum torque or for fast failure detection a quality control of the Diesel combustion process is very important.

Thus, in this paper a method is presented which allows to obtain characteristics of the combustion process without knowledge of the exact combustion heat curve. Cylinder pressure data is required, but computational costs are drastically reduced.

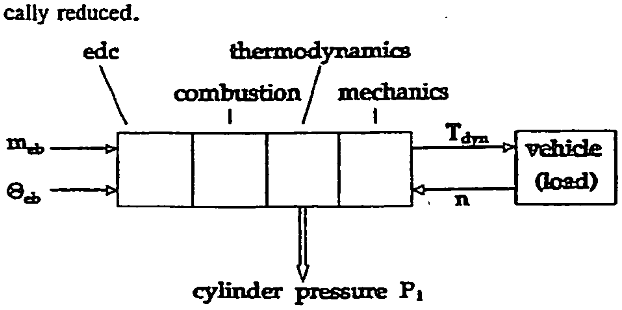

Fig. 1: the Diesel engine as a system

A Diesel engine may be modelled by inputs, outputs and internal states. Injection mass m_{EB} and injection angle θ_{EB} are input variables while the dynamic torque T_{dyn} is the main output signal. The engine speed n is determined by the load dynamics and thus not an independent variable. Internally, the engine can be separated into 4 blocks: the electronic diesel control (edc) unit, the dynamics of combustion, the thermodynamic block and, finally, the kinematics (geometry). In the notation of system theory, the cylinder pressure signals p_i represent internal states, see fig. 1.

When connected to a vehicle, a Diesel engine should actually be viewed as a static actuator since its internal dynamics are neglectable compared to the dynamics of the load (rotational dynamics of the drive shaft, etc.).

2. Analysis of Cylinder Pressure Signals

The combustion in a Diesel engine is a very complicated, nonlinear and stochastic process. A real time analysis of the combustion heat curve would be a good way to determine the motor condition. As a matter of fact, the centre of gravity of the combustion heat curve area depends on the maximum pressure P_{max}, the maximum increase of pressure $(dp/d\Theta)_{max}$ and the maximum gas temperature T_{max}. By sampling the cylinder pressure and solving for the heat transfer, it is principally possible to extract proper combustion quality symptoms. To calculate the transformed energy per rotation angle, one has to solve eq.(1) by integration, see [5]:

$$\frac{dQ_b}{d\Theta}=H_u\frac{\frac{dQ_w}{d\theta}+p\frac{dV}{d\theta}\left[1+\frac{1}{R}\left(\frac{\partial u}{\partial T}\right)_{p,\lambda}\right]+\frac{V}{R}\left(\frac{\partial u}{\partial T}\right)_{p,\lambda}\frac{dp}{d\theta}}{H_u-u+T\left(\frac{\partial u}{\partial T}\right)_{p,\lambda}+\frac{m-B-B_{R_0}}{(B+B_{R_0})^2L_{st}}\,m\left(\frac{\partial u}{\partial \lambda_v}\right)_{p,T}} \tag{1}$$

where :

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$dQ_b/d\theta$ combustion heat curve [J/°degree]
$dQ_w/d\theta$ energy transfer to wall [J/°degree]
B fired fuel mass [kg]
B_{RG} fired fuel in residual gas [kg]
u inner specific energy [J/kg]
L_{st} stoichiometric air requirement [kg/kg]
H_u net calorific value [J/kg]
λ_V air for combustion ratio [-]

However, real time conditions are difficult to match for computation of the combustion heat curve due to the high computational costs.

Due to the strong relation between cylinder pressure signal and the combustion heat curve, the next best signal for monitoring is the pressure signal. In the ideal non-firing case, the pressure signal of an engine is almost symmetric to the top dead centre (TDC). Thus, the centre of gravity in the firing case lies close to the axis of symmetry. Its angle as one of the possible symptoms is close to TDC and does not change too much with variation of input variables m_{EB} or θ_{EB}. It would be possible to determine this centre of gravity very accurately by taking the average of many cycles. This, however, can not be done in real-time.

Therefore, in this work another method is presented that uses the difference between the pressure curve in the firing and in the non-firing case. Unfortunately, the non-firing pressure signal is usually not available in a running motor. Thus, one has to either measure this signal in advance and store it in memory or one need to estimate the non-firing case from the pressure signal before and after firing. This second approach seemed more appealing and was used in this work. The difference pressure curve itself is difficult to interpret in real time, though. By mapping it on a set of symptoms, an online data reduction is accomplished.

The Centre of Gravity of a Function

The following graph shows how to compute the centre of gravity of the difference pressure curve.

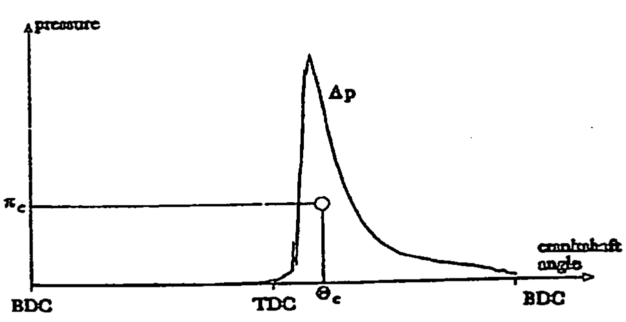

Fig. 2 : Centr f gravity of the pressur curve

Let BDC and TDC be abbreviations f r the Bottom Dead Centre and th Top Dead Centre and let $\Delta p(\theta) = p_{firing}(\theta) - p_{non\ firing}(\theta)$. The coordinates (Θ_c, π_c) of the centre of gravity of the difference pressure curve may be computed by the following equations:

$$\Theta_c = \frac{\int_{BDC}^{TDC} \theta\, \Delta p\, d\theta}{\int_{BDC}^{TDC} \Delta p\, d\theta} \tag{2}$$

where Θ_c is the crank shaft angle of the centre of gravity and

$$\pi_c = \frac{1}{2}\frac{\int_{BDC}^{TDC} \Delta p^2\, d\theta}{\int_{BDC}^{TDC} \Delta p\, d\theta} \tag{3}$$

is the pressure at this point. These two values are strong characteristics for the examination of the combustion. For monitoring the efficiency of the engine the generated torque is important, too.

Difference Pressure Signal Characteristics

Figure 3 shows the geometric arrangement of an idealistic motor with the most important forces and angles.

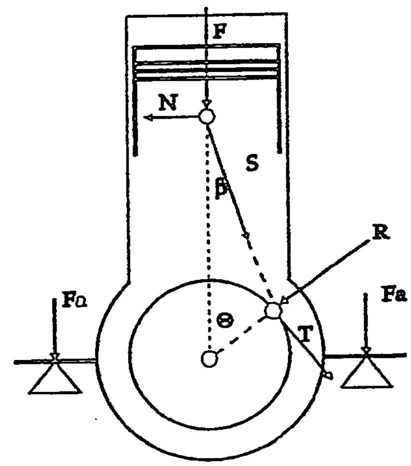

Fig. 3: Forces in a combustion engine

The only interesting parameter for the following conclusions is the cylinder volume V (see [5]). Taking its derivate leads to:

$$\frac{dV}{d\theta} = \frac{V_h}{2}\sin\theta\left[1 + \frac{\lambda}{2}\frac{\cos\theta}{\sqrt{1-\lambda^2\sin^2\theta}}\right] \tag{4}$$

The dynamic torque T_{dyn} generated by the gas pressure $p(\theta)$ is given by the following equation:

$$T_{dyn} = \frac{A\, r}{V_h}\,(p_{(\theta)} - p_0)\,\frac{dV}{d\theta} \tag{5}$$

wh re A is th area of the cross section of the cylinder and p_0 is the environmental pressure. If the torque is integrated ver one cycle, a value proportional to the mechanic work of this cycle is obtained:

$$\Delta W = \int_V p\, dV = \int p \frac{dV}{d\theta}\, d\theta \tag{6}$$

If th difference pressure Δp is multiplied by $dV/d\theta$, a symptom is generated which is proportional to the averaged torque:

$$\Delta T \sim \frac{dV}{d\theta} \Delta p_{(\theta)} \tag{7}$$

The right hand side of this equation is the work which is produced during combustion. If this part is divided by the displacement of the cylinder V_h, the "mean difference pressure Δp_i" is evaluated:

$$\Delta p_i = \frac{\Delta W}{V_c} = \int \Delta p \frac{dV}{d\theta}\, d\theta \tag{8}$$

Δp_i is an average value over one combustion cycle. The resulting torque is directly related to this value and reaches its maximum at the same point. Thus, the following three symptoms for quality supervision of the combustion are available:

Θ_c	The centre of gravity of the combustion
π_c	The height of the centre of gravity
Δp_i	The mean difference pressure

In th following sections is shown how these characteristics are related to the combustion process.

3. Influence of injection parameters

As mentioned above, a Diesel engine has two manipulated inputs: injection angle and injection mass. Between injection and the beginning of the combustion, there is a time delay which should be known. In this case, the combustion process could be controlled by the injection parameters. For example, one c uld drive the motor at maximum torque production. For every engine speed and every injection mass there exists an injection angle for which the transformation of thermal energy to mechanical energy is optimal. For this injection angle the motor reaches its maximum torque.

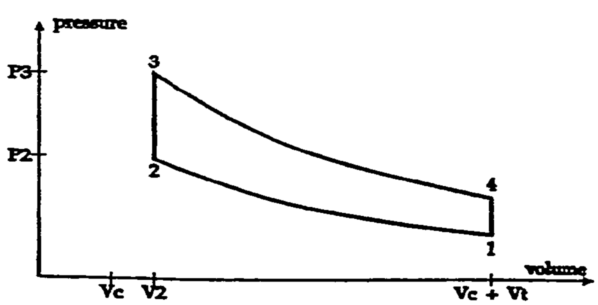

Fig. 4: Constant-Volume-Cycle: P-V-Chart

V_t : displacement f the cylinder
V_c : compression space

To derive a theoretical system model, it is important to know how the symptoms and "height and crank angle of centre of gravity" vary with modification of inputs. To do so, let us assume that the thermodynamical process in a Diesel engine can be modelled by a constant-volume-cycle, see fig. 4. This assumption implies some simplifications to the real case: The time for the combustion process is infinite small and during the compression and expansion of the gas heat transfer doesn't occur.

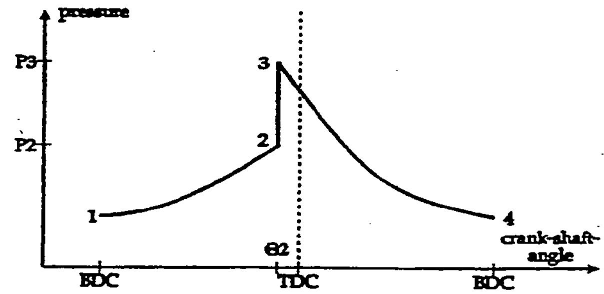

Fig. 5: Constant-Volume-Cycle:
cylinder pressure versus cranc-shaft angle

Figure 5 shows the cylinder pressure for an ideal constant volume cycle. During the adiabatic phases the cylinder pressure is given by the following equations:

$$\begin{aligned} 1 \rightarrow 2: \quad & p = p_1 \left[\frac{V_1}{V}\right]^{\epsilon} \\ 3 \rightarrow 4: \quad & p = p_3 \left[\frac{V_3}{V}\right]^{\epsilon} \end{aligned} \tag{9}$$

With the energy equation $U_3 = U_2 + Q_{23} + H_{C23}$ the pressure at point 3 is given by:

$$\begin{aligned} p_3 &= p_2 + \frac{R}{C_v} (Q_{23} + H_{C23}) \frac{1}{V_2} \\ &= p_1 \left[\frac{V_1}{V_2}\right]^{\epsilon} + \alpha \frac{1}{V_2} \end{aligned} \tag{10}$$

with $\alpha = R/C_v (Q_{23} + H_{C23})$ and H_{C23} the gross caloric value of the fuel. If the difference pressure between the expansion phase and the combustion phase is evaluated, one gets a function which depends only on the cylinder volume V and V_2 (the volume at the beginning of the combustion):

$$\Delta p_{(V,V_2)} = \frac{1}{V^{\epsilon}} \left[\alpha \frac{V_3^{\epsilon}}{V_2} + p_1 V_1^{\epsilon} \left[\frac{V_3}{V_2}\right]^{\epsilon} - p_1 V_1^{\epsilon} \right] \tag{11}$$

Because V_2 is equal to V_3, one gets the following equati n:

$$\Delta p_{(V,V_2)} = \alpha \frac{V_2^{\epsilon-1}}{V^{\epsilon}} = \alpha \frac{1}{V_2} \left[\frac{V_2}{V}\right]^{\epsilon} \tag{12}$$

After integration over V, the height of the centre of gravity is only a function of the volume V_2.

$$\pi_{(V_2)} = \frac{1}{2}\frac{\int_{V_2}^{V_1} \Delta p^2_{(V,V_2)}\, dV}{\int_{V_2}^{V_1} \Delta p_{(V,V_2)}\, dV} \tag{13}$$

Integration may be performed over the volume or the crankshaft-angle. In both cases, the results are the same. If the integration is performed over the volume, one needs to distinguish between two cases:

- ◇ heat transfer before TDC $\quad dV \in [V_2..V_c..V_c+V_h]$
- ◇ heat transfer after TDC $\quad dV \in [V_2..V_c+V_h]$

Heat transfer after TDC is the easier case. Solving eq. (13) leads to:

$$\pi_{(V_2)} = \alpha\,\frac{\kappa-1}{2\kappa-1}\,\frac{1}{V_2}\,\frac{1-\left[\dfrac{V_2}{V_c+V_h}\right]^{2\kappa-1}}{1-\left[\dfrac{V_2}{V_c+V_h}\right]^{\kappa-1}} \tag{14}$$

If heat transfer occurs before the top dead centre, the integral in eq. (13) has to be split up into two parts: From V_2 to V_c and from V_c to $V_1 = V_c + V_h$. For this case the following result is obtained:

$$\pi_{(V_2)}=\alpha\frac{\kappa-1}{2\kappa-1}\frac{1}{V_2}\frac{1-\left[\dfrac{V_2}{V_c+V_h}\right]^{2\kappa-1}-2\left[1-\left[\dfrac{V_2}{V_c}\right]^{2\kappa-1}\right]}{1-\left[\dfrac{V_2}{V_c+V_h}\right]^{\kappa-1}-2\left[1-\left[\dfrac{V_2}{V_c}\right]^{\kappa-1}\right]} \tag{16}$$

Note that for $V_2 = V_c$ eq. (14) and (15) are equal. This is the trivial case: the combustion happens exactly at the top dead centre.

In fig. 6 the height of the centre of gravity π_c is plotted versus the cylinder volume at which combustion occurs.

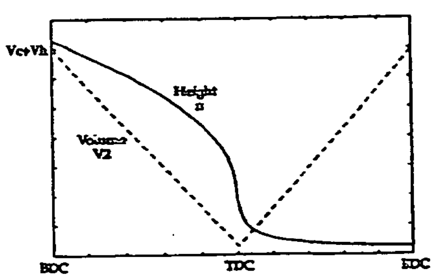

Fig. 6: Height of centr of gravity versus cylinder volume

It is obvious that π_c is a constantly decreasing curve. The function is nonlinear, especially in the interesting area before the TDC. For the crank angle f the centre f gravity, Θ_c, similar relationships can be derived. The centre of gravity moves to later crank angles if combustion occurs later, i. e. it increases with a later injection angle.

The results of these theoretical derivations indicate that π_c and Θ_c are not dependend on output variables like engine speed **n** and should only vary with input signals (injection angle and mass).

4. Experimental results

In this section it will be shown that simulation results from former section can indeed predict the behaviour of an existing motor. The data has taken from a dynamical engine test stand with a Volkswagen Rabbit motor (4 Cylinders, 40 KW, 1.6 l stroke volume).

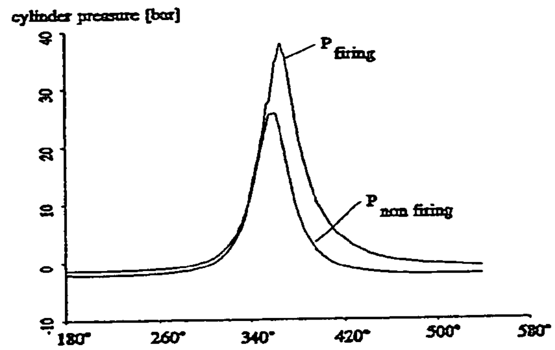

Fig. 7 : cylinder pressure curves, 2400 rpm, full load (measured data)

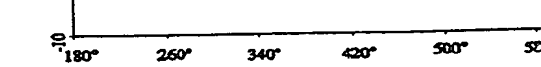

Fig. 8 : difference pressure curve: $P_{firing} - P_{non\ firing}$

Fig. 7 shows measured pressure curves for a firing and a nonfiring engine. The offset between the two curves is caused by sensor drifting and is taken care off during real time computation. The resulting difference pressure curve is shown in fig. 8. Note that by subtracting the non-firing pressure actually the variati n range of the centre of gravity is amplified.

In fig. 9, the height of the centre of gravity is plotted versus the injection angle and the injection mass. As predicted in section 3, this symptom increases with a growing injection mass and with a decreasing injection angle.

In constraint to the theoretical predictions of section

4, Θ_c has a slightly different behaviour. Fig. 10 gives the measured crank angle of the centre of gravity plotted versus the injection parameters.

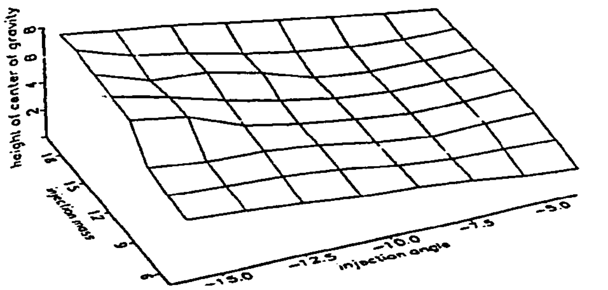

Fig. 9: height of the centre of gravity

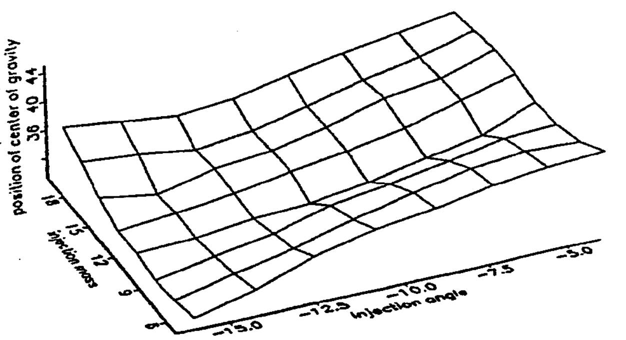

Fig. 10: crank angle of the centre of gravity

Crank angle Θ_c is given in degrees relative to the top dead centre and is a constantly increasing function in direction of the injection angle. Engine speed in fig. 10 and fig. 9 was 2400 rpm.

From the experiments, it seems that in contrast to the theoretical results, the symptoms have a week dependency on engine speed **n**.

5. Classification with neural networks

So far, these results are rather encouraging. Therefor, with a proper classification algorithm it should be possible to map the varying symptoms to a set of faults. The whole process is a pattern recognition algorithm, see fig. 11.

For nonlinear classification, artificial neural nets (ANN) have recently been shown to have desirable properties, see [2]. One of the most widely used algorithm is the feedforward neural net with backpropagation training. As a first attempt, the three symptoms and engine speed n were used as input variables for ANN training. Several configurations of this net were investigated regarding their potential for monitoring and classification of combustion conditions. Configuration and real time application issues are under investigation.

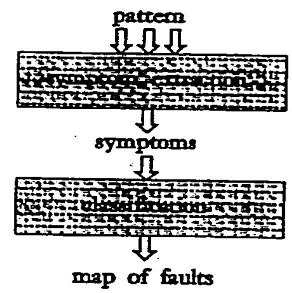

Fig. 11: Principle of pattern recognition

6. Conclusions

The difference pressure signal has been used for real time monitoring and for extraction of motor characteristics. Neural networks seem to be a feasible tool for failure detection in the combustion process, current research deals with optimizing its usability for on board diagnosis systems.

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